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THERMAL AND FLUID DYNAMIC BEHAVIOR OF TRANS-CRITICAL CARBON DIOXIDE HERMETIC RECIPROCATING COMPRESSORS: EXPERIMENTAL INVESTIGATION

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ABSTRACT

An experimental unit has been built to analyse trans-critical carbon dioxide refrigerating cycles in general, and to test CO_2 hermetic reciprocating compressor prototypes in particular. The experimental data has also allowed to numerically compare not only each element, but also all the whole refrigerating system. A description of the numerical compressor prototype is presented and studied in a companion paper. The objective of this work is to show the hermetic reciprocating compressor CL15 that has been built, to work with carbon dioxide as fluid refrigerant under a trans-critical cycle. A complete description of the experimental equipment and the instrumentation used is also included. Finally, the present paper presents extensive comparative experimental results under different working conditions considering low evaporation temperatures, high evaporation temperatures, and different superheating and gas cooler pressures. The results obtained show a promising expectation. Comparative results between the numerical study and the experimental validation of the whole refrigerating cycle is also presented in another companion paper.

1. INTRODUCTION

Since 1992, the CO_2 revival as non-contaminant, non-flammable and unharmful natural fluid refrigerant is an evidence in the technical literature (Lorentzen and Pettersen, 1992),(Lorentzen, 1995),(Frivik and Pettersen, 1997). Nowadays, there are basically two heating and cooling applications where carbon dioxide is a reality: car air-conditioning (Pettersen, 1994), and some heat pumps (Neksa, 1994). However, there is no evidence of the immediate carbon dioxide used in cooling equipment applications, although there is other first hermetic reciprocating compressors prototypes (Yanagisawa et al., 2000), (Fagerli, 1996) or (Süss, 2002) available in the technical literature.

The aim of this work is to present a hermetic reciprocating compressor prototype previously numerically studied and compared in a companion paper (Pérez-Segarra et al., 2004), to describe the experimental unit designed and built to test not only CO_2 hermetic reciprocating compressors, but also experimentally validate the whole trans-critical carbon dioxide cycle (Rigola et al., 2004), and finally to show the experimental results obtained under different working conditions, changing the evaporation temperature, the super-heating temperature or the gas cooler pressure.

The results obtained show a promising expectation that can be improved, if the compressor geometry is numerically optimized by means of hermetic reciprocating compressor numerical models that simulate the thermal and fluid dynamic behaviour (Pérez-Segarra et al., 2003),(Rigola et al., 2003).

2. CL15 COMPRESSOR PROTOTYPE

Figure 1 shows the CL15 prototype shell scheme and perspective to guarantee security high pressure conditions. Table 3 presents an estimation of the main crankcase CL15 compressor prototype parameters: suction and discharge plenum, compression chamber characteristics, and suction and discharge valve geometries.

Figures 2 and 3 show the compressor prototype and its integration in the experimental unit, respectively. The experimental data obtained specifically for the compressor are: inlet and outlet temperature, inlet and outlet pressure, together with mass flow rate and power consumption.

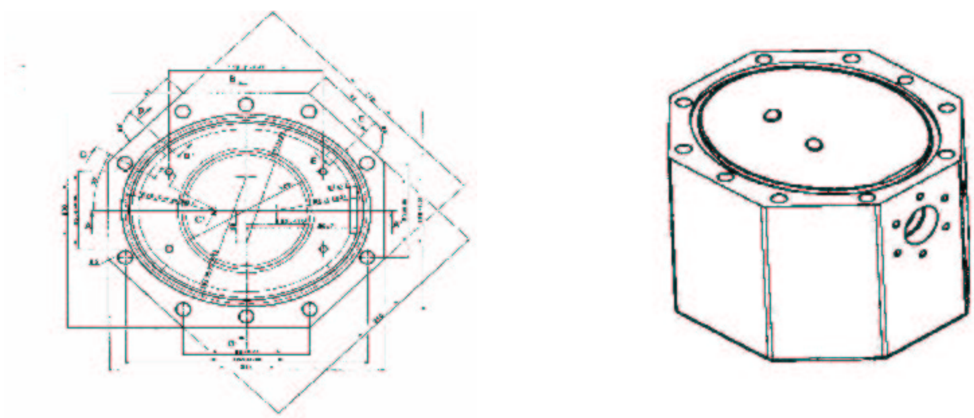

Figure 1: CL15 compressor prototype shell. left: scheme, right: perspective.

Table 1: Crankcase compressor main parameters.

Inlet section diameter	6.2 mm	Outlet section diameter	5.0 mm
Suction line		Discharge line	
2 series chambers	5.08 cm^3	2 series chambers	9.0 cm^3
	1.65 cm^3		6.5 cm^3
plenum suction	1.00 cm^3	plenum discharge	6.6 cm^3
Shell volume	1596 cm^3	clearance ratio	4.38%
Compression chamber		Compression chamber	
bore diameter	14.0 mm	length stroke	9.744 mm
clearance volume	4.38 %	nominal frequency	3000 rpm
suction diameter (1)	3.2 mm	discharge diameter (2)	3.0 mm
suction stop	0.8 mm	discharge stop	0.8 mm
Suction valve geometry		Discharge valve geometry	
suction mass	0.712 g	discharge mass	1.071 g
suction ξ	3.51	discharge ξ	2.33
suction ω	5006.4 Hz	discharge ω	3340.6 Hz

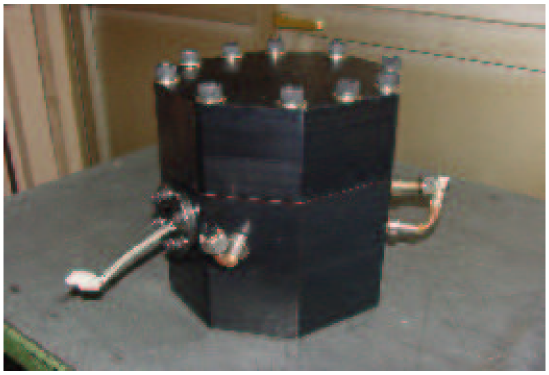

Figure 2: CL15 compressor prototype.

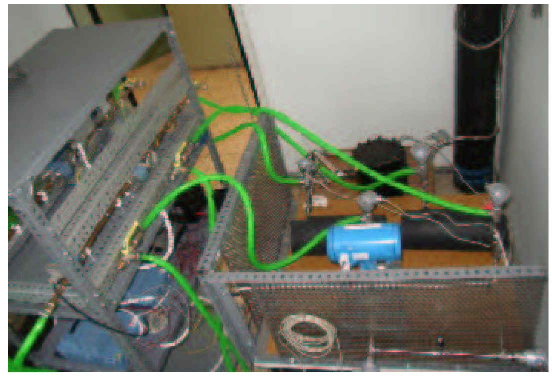

Figure 3: CL15 in the experimental unit.

3. EXPERIMENTAL UNIT DESCRIPTION

Figures 4 and 5 show the experimental unit scheme, together with a lateral view, respectively. The general characteristics of this single stage vapor compression refrigerating system are depicted in Table 2. The experimental units are made up of the following elements: the CL15 carbon dioxide hermetic reciprocating compressor prototype, one double-pipe gas cooler and one double-pipe evaporator together with a metering valve. The auxiliary fluid used in the gas cooler and the evaporator annuli is water.

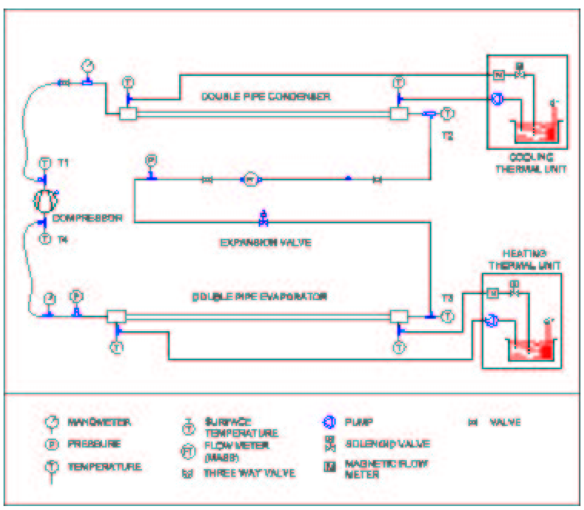

Figure 4: CL15 compressor prototype.

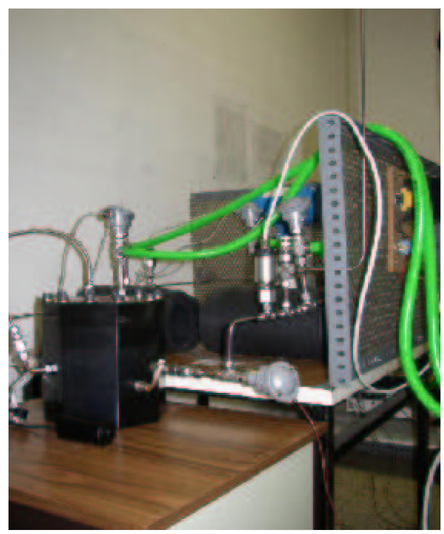

Figure 5: CL15 in the experimental unit.

Table 3 shows the main characteristics of the instrumentation elements. The fluid flow temperatures inside the tubes are measured with K-type thermocouples encapsulated with steel, and electrically insulated by MgO between two wires and the thin steel stick. The fluid flow inside the annuli is measured with platinum resistance Pt-100 thermometer sensors. All temperature points measurements are located at the inlet and outlet cross sections at the inlet of each element of the main and secondary circuits. The temperature accuracy is lower than $\pm 0.03^o$C. The gas cooler and evaporator pressure are measured by pressure transducers with a range from 0 to 150 bars and an accuracy less than 0.1% of span. Refrigerant mass flow rate is measured by means of a coriolis mass flow meter with an accuracy of $\pm$ 0.015% of the measured value. Volumetric flow in the secondary circuits is controlled by two modulating solenoid valves and measured by means of two magnetic flow meters with an accuracy of $\pm$ 0.01 l/min from 0 to 2.5l/min.

Table 2: General characteristics of cycle components.

Tubing		Tube fittings	
Stainless Steel	1/4 OD	Stainless Steel	1/4 OD
Gas cooler		Evaporator	
Dual Heat Transfer Coil		Dual Heat Transfer Coil	
Material	stainless steel	Material	stainless steel
Sample tube	1/4 OD	Sample tube	1/4 OD
External tube	1/2 OD	External tube	1/2 OD
Number of coils	15	Number of coils	15
Insulation thickness	20 mm	Insulation thickness	20
Compressor		Metering valve	
CL15		PARKER	
Cylinder capacity	1.5 cm^3	4Z(A)-NSL-V-SS-V	

Table 3: General characteristics of the instrumentation elements.

Mass flow meter		Security valve	
limits	from 0.3 to 350 kg/h	chocked x_T	0.67
accuracy	± 0.015% measured	C_v	0.41
repeatability	± 0.015 % nominal flow	Spring kit	155-206 bars
stability	± 0.015 % nominal flow		
Temperature sensors Pt100		K-type thermocouples	
deviation	$< 0.1\ ^oC$	deviation	$< 0.4\ ^oC$
accuracy	$\pm\ 0.03\ ^oC$	accuracy	$\pm\ 0.03^oC$
Pressure transducers		Metering valve	
limits	0 - 150 bars	limits	0 - 100 bars
		chocked x_T	0.64
accuracy	< 0.1 % of span	accuracy	< 0.1 % of F.S.
		C_v	0.039

4. EXPERIMENTAL RESULTS

The objective of this section is to present different experimental comparative results, obtained under a range of working conditions to experimentally show the possibilities of this first carbon dioxide compressor prototype (CL15). From these results it is possible to determine the discrepancies and to propose improvements to optimize the CL15 compressor behaviour in other compressor prototypes.

The comparative results obtained have been divided into three main groups. The first group corresponds to four cases with a gas cooler pressure around 100 bars and evaporation temperatures from -10.0^{o}C to 0^{o}C, the super-heating temperature is around 25^{o}C in all these cases. The second group corresponds to three cases with a gas cooler pressure around 100 bars, evaporation temperature from 2^{o}C to 9^{o}C, and super-heating temperatures between 15^{o}C to 25^{o}C. Finally, the third group corresponds to two cases changing the gas cooler pressure from 81 bars to 100 bars.

4.1 Comparative results at low evaporation temperatures.

Table 4 shows the working conditions of each one of the four studied cases, while Table 5 shows the experimental results obtained.

Table 4: Working conditions at low evaporation temperatures.

	T_{sh} [^oC]	p_{ev} [bar]	p_{gc} [bar]	Π [-]	T_{ev} [^oC]	T_{gc} [^oC]
case A	24.28	26.61	102.63	3.857	-10.04	25.31
case B	24.21	30.02	102.64	3.419	-5.72	25.38
case C	24.22	32.85	98.76	3.006	-2.24	25.35
case D	24.14	35.56	101.31	2.849	0.49	25.41

Table 5: Experimental results of each one of the four studied cases.

	$\dot{m}$ [kg/h]	$\dot{W}_e$ [W]	$\dot{Q}_{gc}$ [W]	$\dot{Q}_{ev}$ [W]	η_v [%]	COP [-]	T_{out} [^oC]
case A	5.5	335	420.1	338.9	36.63	1.012	119.44
case B	6.7	350	503.6	404.2	38.51	1.155	116.41
case C	7.3	347	548.0	430.9	37.47	1.242	114.97
case D	9.8	361	726.0	569.3	45.37	1.577	113.26

The experimental results in Table 5 show that the mass flow rate, the power consumption and the COP decrease when the evaporation temperature decreases, while the outlet temperature increases. Differences between the studied case are higher when the evaporation temperatures increases.

4.2 Comparative results at high evaporation temperatures.

Table 6 shows the working conditions of each one of the three studied cases, while Table 7 shows the experimental results obtained.

Table 6: Working conditions at high evaporation temperature.

	T_{sh} [^oC]	p_{ev} [bar]	p_{gc} [bar]	Π [-]	T_{ev} [^oC]	T_{gc} [^oC]
case A	14.8	36.84	98.06	2.662	1.76	29.47
case B	24.11	42.51	102.54	2.412	7.39	29.63
case C	24.07	44.52	100.63	2.260	9.26	29.70

Table 7: Experimental results of each one of the four studied cases.

	$\dot{m}$ [kg/h]	$\dot{W}_e$ [W]	$\dot{Q}_{gc}$ [W]	$\dot{Q}_{ev}$ [W]	η_v [%]	COP [-]	T_{out} [^oC]
case A	10.7	358.0	699.57	536.7	47.23	1.499	99.23
case B	14.0	375.0	926.82	723.0	50.54	1.928	102.79
case C	15.3	228.4	999.6	771.2	51.52	2.101	100.33

This second group of cases present the same tendency than the first group analysed above. From the comparative study between the groups of both sections, it is possible to conclude that in all cases mass flow rate, power consumption and COP decreases when evaporation temperature decreases, while the outlet temperature increases when evaporation temperature decreases.

However, there are two slopes around 0^{o}C, while the slope at evaporation temperatures under 0^{o}C is lower, at evaporation temperatures over 0^{o}C it is higher.

On the other hand, a comparison between case D of the section 4.1 with case A of the present section, with similar evaporation temperature, similar pressure conditions, although with different super-heating temperature, it is possible to conclude that mass flow rate increases when the super-heating temperature increases. On the other hand, the poser consumption, the cooling capacity and the COP decreases, when super-heating temperature decreases.

4.3 Comparative results with an evaporation temperature of 7.2^{o}C, modifying the gas cooler pressure.

Table 8 shows the working conditions of two studied cases, while Table 9 shows the experimental results obtained. Case B in the present section is really the same case B in section 4.2. The idea in these comparative results, considering an evaporation temperature of 7.2^o, is to analyse the influence of the gas cooler pressure.

Table 8: Working conditions for both studied cases.

	T_{sh}	T_{out}	p_{ev}	p_{gc}	Π	T_{ev}	T_{gc}
	[^{o}C]	[^{o}C]	[bar]	[bar]	[-]	[^{o}C]	[^{o}C]
case A	24.0	93.67	42.38	81.21	1.916	7.26	32.02
case B	24.11	102.79	42.51	102.54	2.412	7.39	29.63

Table 9: Experimental results of both studied cases.

	$\dot{m}$	$\dot{W}_e$	$\dot{Q}_{gc}$	$\dot{Q}_{ev}$	η_v	COP	T_{out}
	[kg/h]	[W]	[W]	[W]	[%]	[-]	[^{o}C]
case A	14.85	290	896.12	664.87	53.61	2.293	99.23
case B	14.0	375	926.82	722.97	50.54	1.928	102.79

It is interesting to remark that, under the ideal thermodynamic working conditions (Pérez-Segarra et al., 2004), the COP of case A would be lower than the COP of case B. However, the present section shows that the COP of case A is higher than the COP of case B. In conclusion, the optimum COP consideration from the thermodynamic results would be reviwed.

5. CONCLUSIONS

A first carbon dioxide hermetic reciprocating compressor prototype of 1.5cm^3 has been built and tested in a specific trans-critical CO_2 experimental unit specially designed to control and evaluate this type of compressors under different working conditions. The compressor has been numerically presented in a companion paper (Pérez-Segarra et al., 2004), while a comparative analysis between numerical results and experimental data is also shown in another companion paper (Rigola et al., 2004).

The experimental investigation carried out has allowed to validate the compressor prototype working and to show the first trans-critical cycle tendencies, far from the ideal thermodynamic hypothesis and results, although with a necessity to numerically improve and optimize a second prototype in order to obtain equal and better results than conventional R134a hermetic reciprocating compressors under sub-critical cycle. The results of the first prototype has shown a promising expectation.

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